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General Electric – Alstom merger brings visions of the Überturbine

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Guest Feature

General Electric – Alstom merger brings visions of the Überturbine

Also in this story

By S.C. Gülen, PhD, PE
Principal Engineer, Bechtel Corporation

Integration of the two OEMs' advanced gas turbine designs could deliver combined cycle efficiencies that leapfrog the best available today.

The recent decision by the French government favoring General Electric's acquisition of Alstom's thermal business and subsequent approval of the $17 billion deal by Alstom's Board (a mere formality in light of the board's well publicized endorsement of the deal) opens up the prospect of a new super gas turbine.

Based on old concept

Envisioned is the potential for the integration of GE's steam-cooled turbine technology and Alstom's reheat combustion design to come up with a practical steam-cooled reheat design. The concept of a steam-cooled reheat combustion gas turbine is more than three decades old, e.g. see [1].

Individually the two pillars of the concept, namely steam cooling of hot gas path (HGP) components and reheat (sequential) combustion, have already been deployed in successful commercial units: GE's H-System™ and Alstom's GT24/26 gas turbines, respectively.

The combination of the two technologies has been proposed and analyzed in the past -- always turning out as offering the most efficient combined cycle system [2] possible. This article takes another look at the thermodynamics behind the analysis to quantify the inherent advantage of the concept.

Up to now neither company has given any public indication of actively pursuing the idea as far as the author knows. (In all likelihood they must have looked at it internally as evidenced, for instance, by old ABB patents.) The reason for that is easy to surmise: size, complexity and cost of the overall system. Now that the two companies are merging into one, this Überturbine might finally emerge as a viable commercial product.

In addition to the announced merger, there are two external drivers at play here: 1) ever higher firing temperatures are pushing the limits of dry low NOx (DLN) combustor design to achieve low emissions and 2) increasing need for gas-fired clean base load power generation (relatively speaking) to replace old pre "Clean Air Act" coal-fired clunkers and feared nuclear plants.

At the edge of the NOx barrier

There are three mechanisms for NOx production in the combustor of a gas turbine: thermal, nitrous oxide and prompt NOx – each of which is described by different chemical reaction paths.

Of these three, when flame temperatures are above 2,780°F the dominant mechanism is **thermal NOx** or the **extended Zeldovich** mechanism. Below this temperature, thermal reactions are relatively slow. Beyond about 3,100°F (1,700°C), thermal NOx production grows exponentially (see Figure 1). This can be considered as an upper limit for DLN combustion.

Current advanced H and J class machines with 2,900+°F (1,600°C) turbine inlet temperatures (TIT) operate at the edge of this limit. (Note: combustion flame zone temperatures are higher than TIT.)

Dry low NOx technology can be tweaked to go up in TIT maybe by another 100°F or so. One gas turbine OEM employs **axial fuel staging** (also known as "late lean" injection) to alleviate increased NOx production at high firing temperatures but even that is expected to hit a limit quite soon.

Fig 1. NOx emissions as a function of flame temperature for a typical dry low NOx combustor.

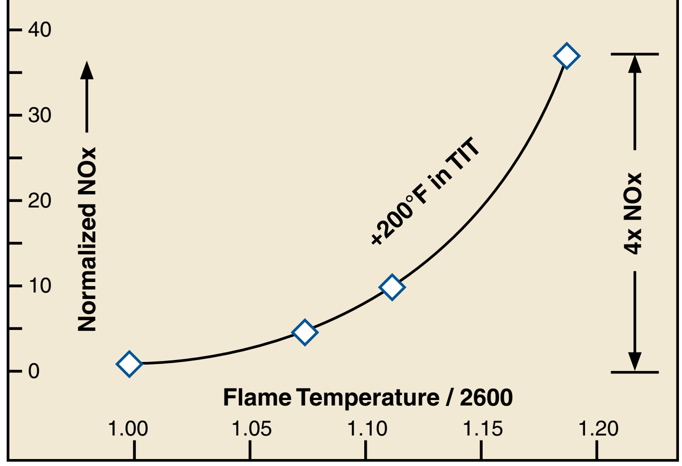
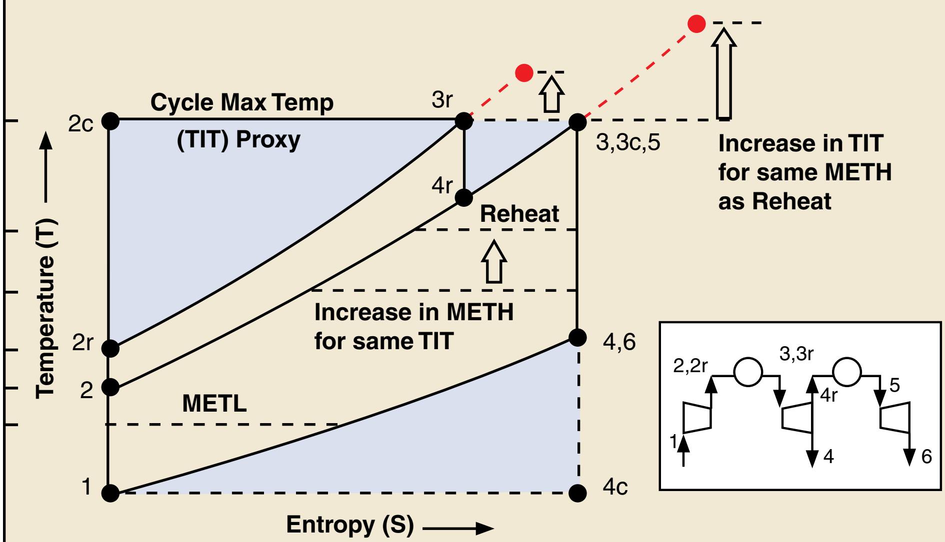


Fig 2. Heat addition and heat rejection irreversibility (losses) of the ideal Brayton cycle (1-2-3-4-1) are represented by the triangular areas (2-2c-3c-2) and (1-4-4c-1), respectively. Reheat cycle (1-2r-3r-4r-5-6-1) reduces the heat addition irreversibility as quantified by the area (2-2r-3r-4r-2). The net effect is an increase in cycle mean-effective heat addition temperature (METH) and cycle efficiency without an accompanying increase in cycle maximum temperature ($T_{3,3r}$).



Another builder looking into ~3,100°F (1,700°C) class gas turbines had to consider exhaust gas recirculation (EGR) for NOx control, which adds significant cost and complexity to the design.

Thus, the only surefire way to keep NOx emission in check is to rein in the urge to go full blast with turbine inlet temperature and not sacrifice efficiency. This is where the reheat combustion concept enters the picture.

Reheat Gas Turbine

It is a well-known axiom of thermodynamics that does not hurt repeating: "Any heat engine cycle is a valiant albeit vain attempt to replicate the Carnot cycle".

The biggest hurdle in this somewhat quixotic engineering quest is achievement of isothermal heat transfer. Reheat or sequential combustion is a modest approximation of isothermal heat addition, which can be found in any undergraduate textbook.

The goal is to realize an increase in the cycle effective heat addition temperature (see Figure 2) without increasing the turbine inlet temperature which is the maximum cycle temperature. For a fundamental discussion of reheat, ideal cycle efficiency can be written as

$$\eta = 1 - \frac{METL}{METH}$$

where METL and METH are the cycle's mean-effective heat rejection and heat addition temperatures. (They are logarithmic averages of heat transfer beginning and ending temperatures; i.e., T_2 and T_3 for METH and T_4 and T_1 for METL.)

In passing, a commonly encountered mistake is to confuse ideal cycle efficiency with the "ultimate" Carnot efficiency, $1 - T_1/T_3$, which represents the theoretical maximum.

The key message here is that in order to have the same mean-effective heat addition temperature (that is, the same cycle efficiency) as the reheat cycle in Figure 2, a non-reheat cycle must increase its cycle maximum temperature (a proxy for TIT) and/or cycle pressure ratio.

As illustrated by Figure 2, the increase would be much higher at the non-reheat cycle's pressure ratio (P_2/P_1). On an ideal cycle basis, the advantage of reheat cycle over the non-reheat cycle is illustrated in Figure 3.

Also shown in Figure 3 is an estimate of the realistically achievable performance advantage, which is much more modest than predicted by the ideal cycle comparison. The primary drivers for this are increased hot gas path component cooling load and combustor design requirements.

In fact, for turbine inlet temperature values of approximately 1,450°C (2642°F) or above, the reheat cycle efficiency advantage disappears due to significant increase in cooling losses [3]. This is where closed-loop steam cooling concept enters the picture.

Steam cooled gas turbine

For all practical purposes, there is only one closed-loop steam cooled gas turbine: General Electric's H-System.

Fig 3. The shaded box designates the most likely near-to-midterm impact of closed-loop steam cooled reheat gas turbine.

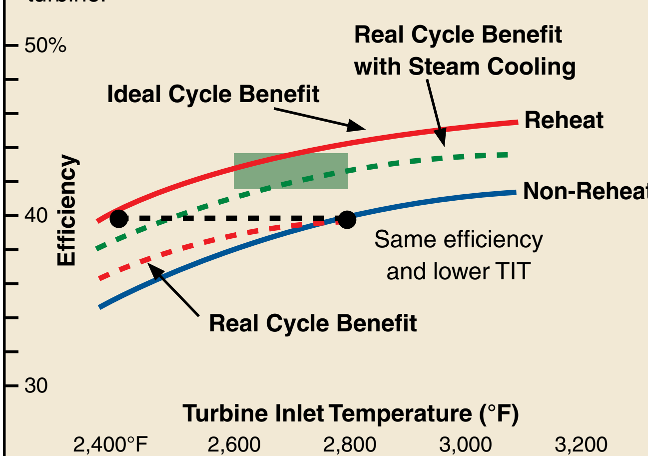
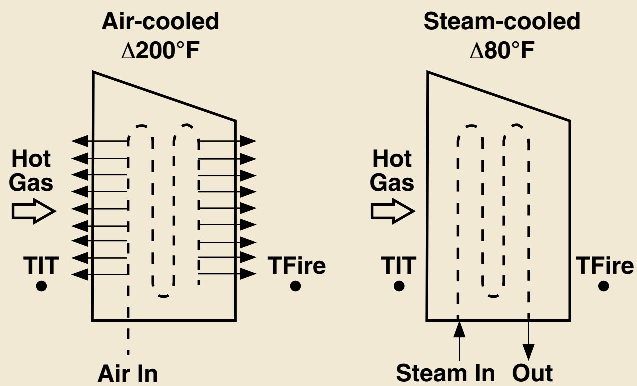


Fig 4. Internal air cooling reduces firing temperature into stage 1 turbine blades by about 200°F versus 80°F with steam cooling.



Admittedly, it is true that Mitsubishi G and J class gas turbines also employ steam cooling for combustor liner, transition piece and stage 1 and 2 turbine rotor rings (J class).

In terms of hot gas path "chargeable" and "non-chargeable" cooling air reduction, however, Mitsubishi's G and J class gas turbines are essentially air-cooled machines.

(It should be noted that Mitsubishi did design and test a fully steam cooled "H" machine almost 15 years ago, back around 2000-01, which had a cycle pressure ratio of 25 to 1. It was never offered commercially but its compressor design lives on in current G and J class gas turbines.)

In H-System gas turbines, on the other hand, closed-loop steam cooling reduces hot gas temperature drop across the stage 1 nozzle to less than 80°F.

For the same combustor temperature and turbine inlet temperature, this results in an increase of 100 to 150°F in firing temperature vis-à-vis advanced F class machines with air cooling (Siemens H class gas turbines also belong in this category).

An additional benefit of steam cooling is less parasitic extraction of compressor discharge air and higher flow to the head-end of the dry low NOx combustor for fuel premixing. If the firing temperature is kept at the F class level, the benefit of steam cooling presents itself as reduced turbine inlet and combustor temperatures, i.e., reduced NOx production.

In the H-System, the first two turbine stages are fully steam cooled including nozzles and buckets. This reduces the amount of chargeable cooling air and increases gas turbine output via higher gas flow through the hot gas path.

Heat rejected to the coolant steam is converted into additional steam turbine power output. The net benefit of full steam cooling is a two percentage points increase in combined cycle efficiency [2,3].

Air cooling also needed

Closed-loop steam cooling does not eliminate air cooling altogether. Air purging is still needed to prevent ingestion of hot gas into the wheel spaces.

In addition, air is used for cooling the trailing edges of stage 1 and 2 nozzle vanes via internal coolant flow (presents a challenge). Supplementary cooling of inner and outer side walls (platforms) and trailing edge of the nozzle vanes with wheel space purge air is also a requisite to ensure adequate parts life.

Steam cooled gas turbines, of necessity, are only available in combined cycle configuration. In fact, they are more aptly described as "integrated steam/gas" cycles [1].

The connection between the topping and bottoming cycles goes way beyond the exhaust gas duct between the gas turbine and heat recovery boiler (HRB). The network of alloy pipes and valves required to interconnect them, in addition to a cooling air cooler (kettle reboiler type heat exchanger) for IP steam generation (not to mention the performance enhancing fuel heating), results in a veritable (and expensive) maze.

The cooling air cooler is a consequence of the high Brayton cycle pressure ratio (23 for the H-System) requisite for an optimal design necessitated by high firing temperature (2,600+°F) and reduced hot gas dilution by coolant in the hot gas path (leading to high compressor discharge temperatures).

A significant hurdle in H-System bottoming cycle design is excessive reheater pressure drop (approximately 25% vis-à-vis typical 10-12% for modern reheat steam bottoming cycles) caused by the HGP cooling steam circuit embedded within the reheat steam piping. ■

Part 2

Engineering building blocks for a Überturbine prototype

By S.C. Gülen, PhD, PE
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The modern steam-cooled H-System and the GT 24/26 reheat combustion design represent the two unique gas turbine architectures needed for the Überturbine.

Despite undeniable performance benefits, neither steam cooling nor reheat managed to vanquish their conventional air-cooled rivals, whose basic design has not changed much from the pioneering jet engines of 1940s and 1950s.

As of today there are only six H-System units in commercial operation. Moreover, for quite some time, the H-System has not been offered by GE commercially – even though listed in GE's product portfolio.

Most recently, GE announced the air-cooled HA class machines, which draw heavily upon the technologies proven in their steam cooled predecessors (e.g., single crystal materials, advanced thermal barrier coatings and 4-stage turbine section).

As far as reheat combustion is concerned, there are many more GT24/26 units in commercial operation. Nevertheless today's owner of the technology, Alstom, fell way behind the leading OEMs in terms of worldwide gas turbine sales. Why is that? A short review of the history behind the current design provides an answer.

Reheat gas turbine background

The idea of reheat or sequential combustion has been around for quite a long time. Stodola explicitly referred to it as "a means to increase efficiency" in an article he wrote right after he oversaw performance testing in 1939 of the world's first industrial gas turbine [4].

Brown Boveri Corp (BBC) developed the concept into working hardware and, in 1948, built and tested two such gas turbines in Beznau, Switzerland. These machines were quite different from the compact "jet engines on steroids" that one tends to associate with the term 'industrial gas turbine" these days.

They were rather complex power plants in their own right with an intercooled two-shaft configuration comprising separate low pressure (LP) and high pressure (HP) compressor-turbine trains and large external single-can combustors.

In the 1950s, BBC supplied such "tailor made" units all over the world, including 4 x 25 MW for the Port Mann station in Vancouver, BC and a single-unit plant in Lima, Peru. Another more recent and well known site is the Huntorf compressed air energy storage plant in Germany with its single-shaft HP-IP turbine and two silo combustors.

Asea Brown Boveri, descendant of the venerable BBC company, took an evolutionary design path in 1993 with the introduction of a compact GT24/26 (60/50Hz) reheat gas turbine with two annular combustors comprising proprietary EV and SEV burners. (Initial designs included an intercooler which significantly added to engine length and was dropped from the final production design.)

At the time ABB, like other gas turbine OEM suppliers in the industry, did not have an in-house test facility large enough to put the entire machine through its paces prior to customer shipment. With so many innovations involved in the design, this put the first units placed in service at considerably higher risk than usual for the introduction of any new engine.

The first commercial GT24 unit, installed by Jersey Central Power & Light at the Gilbert Station in New Jersey, underwent extensive field trials and prototype testing prior to its operation. In spite of this, the initial series of production units were beset with serious technical problems – largely due to the new 30:1 compressor (with about twice the pressure ratio of existing heavy frame industrial gas turbine compressors) as well as the sequential combustion system.

At first, ABB managed to keep a lid on the field problems and continued to have success in selling new orders well into the pre-2000 boom years. As a result, the promising new technology suffered severe damage to its reputation that would remain for years to come.

Ultimately, ABB terminated further deliveries, allowed orders to be cancelled, compensated clients for damages and devoted large resources to fixing the problems.

In 1999-2000, Alstom formed a joint venture with ABB and subsequently acquired a 50% share of their gas turbine business. Since then Alstom has been the OEM supplier for reheat combustion GT24/26 technologies.

In a 2000 press release Alstom acknowledged the severity of the design issues and field problems associated with GT24/26 and said it was setting aside close to 1 billion Euros to address those issues. Since then, it is fair to say that

GT24/26 reheat gas turbines have established themselves as reliable and efficient power generation systems.

Steam cooling trial and error

The history of component cooling using water or steam goes even further back than the reheat concept.

In 1903, Aegidius Elling patented a gas turbine that included water cooling to lower the hot combustion gas temperature to about 750°F (400°C) at the turbine inlet. The steam generated during the process was mixed with the gas and expanded in the turbine. In essence, it demonstrated a "poor man's H-System" with open-loop steam cooling configuration a century before first fire of GE's 9H at Baglan Bay.

In 1930 Brown Boveri introduced a prototype of Holzwarth's "explosion" turbine (constant volume combustion) which had an inlet gas temperature of about 1300°F (700°C) and water-cooled first stage [4]. And in the 1950s, Siemens invested considerably in the design of a turbine rotor with water-cooled blades for 1800°F (~1,000°C) inlet temperature [5].

The steam generated inside the water-cooled blades was routed out through the hollow rotor and piping. A myriad problems surfaced but were resolved (vibration, water filter clogging and parts overheating) to achieve a turbine inlet temperature of 1930°F (1,055°C) in the tests. But the program eventually folded due to cost issues.

Reheat gas turbine hall. Brown Boveri's two-shaft intercooled, reheat gas turbine power plant in Port Mann, Vancouver, BC, Canada. Note the four 25MW units lined up in a row along the turbine hall. The first unit is shown in the foreground with the generator connected to the low pressure train on the right and high pressure train on the left. The two vertical cylinders on the left are the LP and HP combustors.

Starting in the mid-1970s, GE investigated water-cooled stage one nozzles as part of U.S. DOE's High Temperature Turbine Technology (HTTT) program. Parts were designed and cascade tested in gas temperatures at temperatures of up to approximately 3000°F (1,650°C), the DOE program goal, at 145 psia [6]. Rig tests in an actual turbine similar to a Frame 6 were planned but never carried out.

Difficulties with controlling water-steam phase changes and instabilities associated with nucleate boiling as well as limited coolant temperatures eliminated water as a turbine coolant once and for all. By the time GE joined DOE's Advanced Turbine Systems (ATS) program, closed-loop steam cooling was the chosen path and led to the commercialization of the H-System.

H-System success

The H-System did not run into the same problems and none of the six units in field operation revealed any design flaws. This fact can be attributed to the cautious path that GE took in developing the highly complex design over a period of 10 years, albeit at an exorbitant cost partially offset by DOE's ATS program funding.

Comprehensive testing of the first 109H single-shaft combined cycle power plant in 2003-2004 fully demonstrated the capability of the machine to start in air-cooled mode, transition into steam cooling to reach base load, run as predicted over its entire operating envelope for an extended period, and shut down. (Full disclosure: the author was a GE engineer at the time and participated in the test program.)

The same was true of the other five H-System units (three 50Hz 109H units in Japan and two 60 Hz units in California). Today, the six H-System units have accumulated more than 175,000 fired hours at firing temperatures well above 2,600°F (1,430°C), a level only recently achieved by Mitsubishi's J class gas turbines with 2,912°F (1,600°C) turbine inlet temperature.

The two 107H units at the Inland Empire Energy Center in California, which entered service in 2008, made the top 20 list in heat rate in Electric Light & Power magazine's annual "Operating Performance Ratings for Top 20 Power Plants" articles.

Even though the capacity factor was only about 60%, this is not a bad feat. Furthermore, the Inland plant ranked number one in 2011 and 2012 in terms of NOx emissions rate (0.00385 lbs/MMBtu in 2012).

The two units were successfully tested in the summer of 2008 (the author was there as well) operating with a unique fuel moisturization system for improved efficiency and NOx control. Unfortunately, near the end of the testing in 2008, Unit 2 suffered a compressor failure.

Although never publicly disclosed, the rumored cause of the failure was a manufacturing defect in the compressor's last stages, and the restart was delayed until 2010 due to difficulties encountered in procuring replacement parts. ■

Part 3

Looking beyond air cooling for 64 or 65% net efficiency

By S.C. Gülen, PhD, PE
Principal Engineer, Bechtel Corporation

Gas turbine OEMs are claiming over 61% net efficiency for advanced combined cycle plants. How much higher can steam cooling and reheat realistically achieve?

The previous discussion of engineering building blocks makes the point that, separately, both steam cooling and reheat have been proven reliable in commercial service and capable of delivering superior performance. Their possible use for combined cycle design remains to be seen.

Today, four major OEMs (soon to be only three) make claim to over 60% net combined cycle efficiency for production plants using advanced air-cooling designs; actually GE and Mitsubishi claim better than 61% net efficiency for their HA and J class gas turbine combined cycle plants.

Put aside for a moment the fact that only Siemens has actually "walked the walk" albeit while employing an advanced steam bottoming cycle and taking advantage of ideal site conditions. And let us examine the underlying fundamentals behind combined cycle efficiency and the potential for going beyond advanced air cooling techniques with a "super turbine" employing both steam cooling and reheat combustion.

The combined cycle efficiency can be estimated reasonably accurately as follows:

$$\eta_{CC} = \left(\eta_{GT} + \varepsilon \cdot \left(1-\eta_{GT}\right) \cdot \eta''_{BC}\right) \cdot \left(1-\alpha\right)$$

where η_{GT} is the GT efficiency, ε is the GT exhaust exergy as a fraction of exhaust energy, η''_{BC} is gross bottoming cycle exergetic efficiency which is the ratio of steam turbine generator output to gas turbine exhaust exergy, and α is the plant auxiliary load as a fraction of gross combined cycle output (see Gülen and Smith [7]).

Exergy is the maximum work potential of the working fluid (in this case, gas turbine exhaust gas) of given pressure, temperature and composition. It can only be achieved in a hypothetical Carnot cycle.

With a known equation of state (e.g., JANAF tables for gases) the exergy of a given fluid can be exactly calculated. For a gas turbine exhaust temperature range of 1,100-1,200°F, ε is 0.46-0.48. In other words, maximum work potential of a gas turbine bottoming cycle is roughly 50% of the exhaust gas energy.

A real cycle can feasibly achieve only a fraction of the maximum work potential (the Carnot factor). For the Rankine steam bottoming cycle of a gas turbine combined cycle plant design, this value (η''_{BC} in the formula) is typically around 0.74-0.75 for a 3-pressure reheat steam cycle with steam temperatures 1,050-1,100°F, condenser pressure of 1.2 inches of mercury and an advanced steam turbine with suitably large exhaust annulus.

The value of α, the percent of auxiliary load losses, is 1.6% for typical combined cycle performance ratings listed in the Gas Turbine World Handbook. This is commensurate with once-through open-loop water cooled condenser operation at 1.2 inches of mercury and no fuel gas compression.

With appropriate values for the variables of ε, η''_{BC} and α thereby established, the simple CC efficiency equation lays out the combined cycle vs. gas turbine efficiency landscape concisely, as shown in Figure 6. The takeaways from the figure can be summarized as follows.

• For 60% combined cycle efficiency, minimum 39% gas turbine efficiency, high exhaust temperature (implying system level optimization to determine gas turbine firing temperature and cycle pressure ratio) and a state-of-the-art bottoming cycle are requisite.[1]

• For 40%-plus efficiency gas turbines, over 60% combined cycle efficiency should be eminently achievable. (Caveat: This statement is true only with favorable site conditions suitable to low steam turbine back pressures with minimal parasitic power consumptions.) All bets are off with extremes such as air-cooled condensers in desert climates and/or high site elevations. (The reader is referred to the article by Maher Elmasri in GTW July-August 2013 issue for more on this.)

• Between 40% and 41% gas turbine efficiency, over 61% combined cycle efficiency is a stretch, but possible, given a truly advanced steam cycle and steam turbine. The ability to

1 Note that the reheat gas turbine with open-loop steam cooling proposed by Rice in his 1982 paper [1] had an efficiency of 42.5% and 1,299°F exhaust temperature. It was a bona fide 61+% net GTCC enabler. Unfortunately, Rice was not as visionary with his choice of bottoming cycle (he had a two-pressure cycle with 300°F HRSG stack and feedwater heating) and ended up projecting well below 60% efficiency.

draw cooling water year round from the cold Danube would not hurt either (as is Siemens' good fortune at the Irsching 8000H plant).

• With more than 41% simple cycle efficiency gas turbines, over 61% combined cycle efficiency becomes a realistic prospect.

Current-production F, G, H and J class gas turbines are primarily air-cooled machines, whose performance (over 40% simple cycle efficiency) is driven by high firing temperatures and commensurate cycle pressure ratios (20 to 23) complemented by advanced steam Rankine bottoming cycles to achieve over 60% combined cycle efficiency.

Even more advanced air-cooled designs on the horizon can establish the basis for the best-case scenario air-cooled machines (see Table 1). Their embedded technologies such as advanced aero design, new hot gas path materials and coatings, advanced film cooling techniques and higher component efficiencies can all be retained in a steam-cooled reheat combustion architecture.

Game changing technology

How would steam cooling and reheat change this picture? The improvement obtainable from a closed loop steam cooled reheat configuration is summarized by Table 2.

For full steam cooling, à la General Electric's H-System, from 2 to 2.5 percentage points can be added to combined cycle efficiency when operating at the same TIT conditions and reach 63% to 64% CC efficiency [2,3].

With only stage 1 nozzle steam cooling the adder is about halved to 1 to 1.25 percentage points to operate at 62% to 63% combined cycle efficiency [3] as defined by the green shaded rectangle in Figure 6.

Unless materials significantly reducing the need for HGP component cooling such as ceramic matrix composites come onto the scene in a timely manner, steam cooled reheat technology is the most likely candidate to reach the 65% barrier (or come closest) without running into combustion and emissions problems.

The estimated stage-by-stage ratings for steam cooling with reheat were calculated using Thermoflex software (developed by Thermoflow) based on best "state-of-the-art" air-cooled gas turbine performance. The deltas shown should be considered as purely thermodynamic entitlement values.

Single-stage HP turbine (pressure ratio of 2) and four-stage LP turbine are assumed. The two cases assume two different levels of steam cooling. The first is for steam cooling HP and LP stage 1 nozzle vanes only; the second is for "full steam cooling" of the HP nozzle vanes plus LP stage 1 and 2 vanes and buckets.

Table 1. Composite rating for "best" air-cooled F, G, H and J-class gas turbine design performance.

Gas Turbine Design Parameter	Best Case
Gas turbine output (60/50 Hz)	300-500 MW
Approximate gas turbine efficiency	41+ %
Compressor pressure ratio	22 to 23
Turbine inlet temperature	1,600°C
Turbine inlet temperature	2,912°F
Approximate GT exhaust temp	1145°F
Net combined cycle efficiency	61+ %

For the latter, performance is estimated at two compressor pressure ratios – with the higher value expected to be representative of a final optimized design. Cooling steam is supplied from the cold reheat and returned to the hot reheat line. Cooling air cooler heat rejection is used for IP steam generation in a kettle reboiler.

Bottoming cycle calculations assume state-of-art, three-pressure reheat steam cycle and advanced steam turbine with water-cooled (open loop once-through) condenser. Firing temperature is defined as the rotor/bucket inlet stagnation temperature, and the HGP total cooling airflow rate is expressed as a percentage of compressor inlet airflow.

Can they get there?

GE and Alstom have significant gas turbine architecture differences: i.e., can-annular versus annular combustors and

Fig 6. Combined cycle efficiency as a function of gas turbine efficiency (ISO base with optimal bottoming cycle heat rejection). The two levels of exhaust temperature represent low and high ends of F, G, H and J class heavy frame turbines. Base η_{BC} is 74% for Texh = 1,125°F and 74.7% for Texh = 1,175°F at ISO conditions.

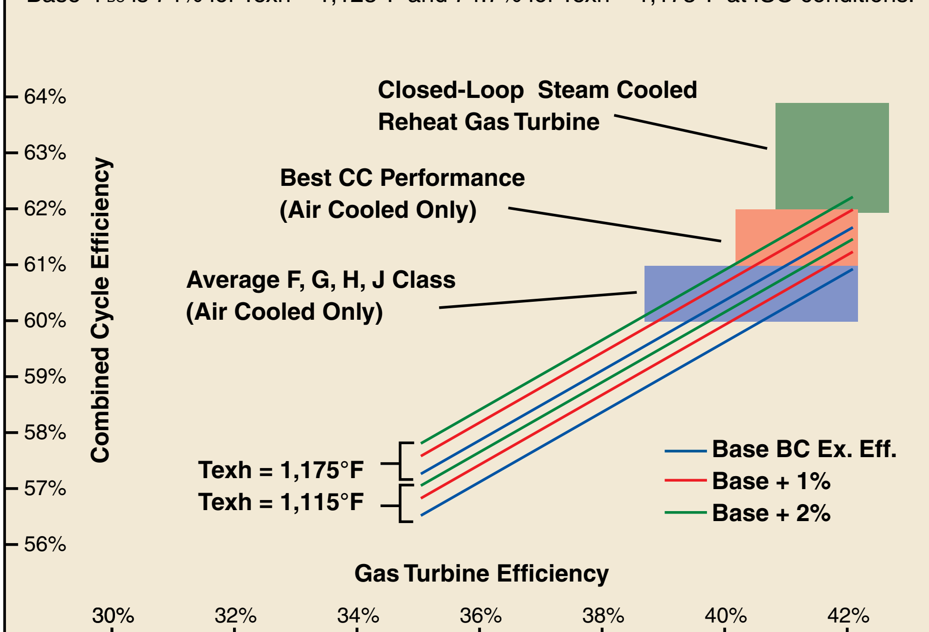

Table 2. Estimated benefit of reheat with steam cooling (two cases) as referred to "best case" air cooling technology, with hot gas path (HGP) total cooling air flow expressed as a percentage of compressor inlet airflow.

Design parameter	Air Cooled	*Only S1N Steam Cooling	**Full Steam Intro Design	**Full Steam Optimized
Reheat combustion	No	Yes	Yes	Yes
Gas turbine output	Base	+ 15%	+ 35%	+ 35%
GT efficiency (points)	Base	+ 0.25	+ 1.30	+ 2.00
Compressor pressure ratio	22.5	35.6	35.6	39.3
HP firing temperature	2,715°F	2,545°F	2,545°F	2,545°F
LP firing temperature	N/A	2,725°F	2,725°F	2,725°F
HGP cooling air flow	28.8%	26.5%	16.4%	16.2%
Exhaust temperature	Base	+ 90°F	+ 180°F	+ 145°F
Cooling air-cooler duty	N/A	5,500 Btu/sec	6,200 Btu/sec	7,250 Btu/sec
Steam cooling duty	N/A	13,590 Btu/sec	23,500 Btu/sec	23,500 Btu/sec
Steam turbine output	Base	+ 22%	+ 40%	+ 35%
Combined cycle net output	Base	+ 19%	+ 38%	+ 35%
CC net efficiency (points)	Base	+ 1.25	+ 2.5	+ 2.75

*Limited steam cooled HP an LP Stage 1 nozzle vanes only

**Fully steam cooled HP nozzle vanes and LP stages 1 and 2 (vanes and buckets)

bolted versus welded disk rotor construction, respectively. The exact nature of technology "osmosis" or "integration" between the two merged organizations remains to be seen.

As far as a potential steam-cooled reheat machine is concerned, the most likely approach is to keep the current GT24/GT26 architecture and integrate the proven cooling steam delivery system into the welded rotor design. Even though the performance entitlement offered by a "fully steam cooled" turbine is highly tempting, the expectation is that cost and complexity issues will preclude it – at least for the next 5 to 10 years. However, steam cooled HP and LP turbine inlet nozzle vanes provide most of the proverbial "bang for the buck" and should be eminently do-able with reasonable investment cost and engineering effort.

Conceivably, the first Alstom (EV) annular combustor can be replaced by a can-annular GE design with axial fuel staging to get the highest possible HP turbine inlet temperature. In all likelihood, however, the second (SEV) annular combustor would be retained for the most compact final configuration.

There is no doubt that a steam-cooled reheat combustion integrated cycle power plant will be quite expensive. But the plant would be more flexible than existing public opinion suggests; it would retain the low-load capability of existing reheat machines and would not be too sluggish in terms of warm/cold starts and load ramping. True, it would not be as nimble as an air cooled "fast start" unit, readily amenable to daily two-cycled load following and/or stand-by. Then again, this is not the intended application for a highly efficient and pricey system most suitable to base load duty.

In conclusion, do not hold your breath but do not totally dismiss a near future announcement of this highly integrated system either. ■

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